BULETINUL INSTITUTULUI POLITEHNIC DIN IAŞI
Publicat de
Universitatea Tehnică „Gheorghe Asachi" din Iaşi
Volumul 69 (73), Numărul 4, 2023
Secţia
CONSTRUCŢII DE MAŞINI
DOI:10.2478/bipcm-2023-0031

INVESTIGATION ON THE PERFORMANCE OF A SECONDARY BRAKE

BY

NARCIS-IOAN ȚICA and DANIELA POPESCU*

"Gheorghe Asachi" Technical University of Iaşi, Faculty of Machines Manufacturing and Industrial Management, 59A Prof. D. Mangeron Blvd., 700050, Iaşi, Romania

Received: December 30, 2023
Accepted for publication: January 27, 2024

Abstract. In order to increase the safety of vehicles, the braking systems have two different braking sequences: the service brake and the secondary brake. The secondary brake is used when a component of the braking system fails to work adequately and it uses only two main components of the brake system, the calipers and the master cylinder. This paper presents a method of investigation, based on Matlab Simulink, to be used to verify the efficiency of the secondary brake by studying the dynamics behaviour, brake bias, stability and control during braking. The outputs are discussed by having in view, the ECE R13 regulations, for a vehicle with the initial speed at v = 60 km/h, mass m = 2300 kg, rolling radius r = 0.32 m, master cylinder mc = 22.2 mm, pedal ratio 4.3 and static friction coefficient 0.9. The results confirm that the variation of yaw rate and slip ratio have a minor influence on the stability and control.

Keywords: secondary brake, ECE-R13 regulations, vehicle dynamics, wheel dynamics.

---

*Corresponding author; *e-mail*: daniela_popescu@tuiasi.ro

1. Introduction

The brake system is an important part of a vehicle, which must be controlled easily by the driver, to stop or to reduce the speed efficiently. The main components of the brake system are the master cylinder, the anti-lock brake system (ABS), the actuator and the calipers. To save space inside the vehicle, new solutions propose one box systems, meaning that the ABS system and the master cylinder are integrated in one box. The advantages are important, such as less components and better answer to transducers, while the performance is same or even better comparing to previous designed solutions.

According to ECE R13 regulations (European Commission, 2016), the brake must have two major sequences of braking, to comply to the safety requirements. One is called service braking and the other one is the secondary braking. The service braking have a high-performance hydraulic power, working at high pressure that generates a high force on the wheel, to safely stop the car within a short distance, without any negative impact on the control of the vehicle and its functionality. The main role of the secondary brake is to replace the service braking, when one component of the brake system fails to work adequately. The secondary-brake system does not have to be an entirely separate third braking system (in addition to the service brakes and the parking brake) with its own separate actuation device. It can consist of the remaining intact circuit of a dual-circuit service-brake system on which one circuit has failed, or it can be a parking-brake that is capable of graduated application. The secondary brake works by using the master cylinder and the calipers, having different configurations. The most common are the two diagonal branches (X configuration) and the an axle (configuration ||). Its performance is lower comparing to the service braking. In the case study presented in this paper, all the calipers will be actuated by the master cylinder (Reza, 2008; Lei *et al*., 2023; Hans, 2006; Dieter, 2014).

This paper presents the methodology to be followed to study the performance of a secondary braking, using Matlab&Simulink software (Mathworks, 2023). The results are discussed by having in view, the ECE R13 regulations, for a Volkswagen Passat vehicle.

2. Methodology to investigate the performance of a secondary brake

To solve the dynamics equations of the vehicle and to investigate the performance of secondary brake, the methodology to be applied uses Matlab& Simulink software. In Fig. 2, the methodology for evaluating the performance of a secondary brake, in accordance with ECE R13 regulations, is presented step by step. In the following, the stopping distance, the deceleration value to keep the vehicle stable enough and easy to be controlled by the driver, the slip ratio and the yaw rate are estimated for a case study, as an example.

The slip ratio is the difference between the theoretically calculated forward speed based on the angular speed of the rim and the rolling radius, and the actual speed of the vehicle, expressed in Eq. (1) as a percentage of the later,

$$\lambda = (v - r\cdot\omega)/v\ , \tag{1}$$

where r is the rolling radius of the wheel, ω is the characteristic angular speed of the wheel and v is the characteristic velocity of the vehicle [m/s] (Rajesh, 2012).

The yaw rate is the rotation angle of the vehicle turning in horizontal plane around the vertical axle passing to its own centre of mass (CG), calculated according by Eq. (2):

$$\dot{\psi} = \frac{v_x\cdot\delta_w}{(l_f+l_r)\cdot\left(1+\frac{v_x^2}{v^2}\right)}, \tag{2}$$

where v_x is the velocity in longitudinal direction [m/s], v is the characteristc velocity of the vehicle [m/s], l_f is the distance between front wheels [m], l_r is the distance between rear wheels [m] and δ_w is the steering wheel angle [º] (Aripin *et al.*, 2014). For the case study presented here, the response of the steering wheel angle during braking, is a sinusoidal curve, presented in Fig. 1.

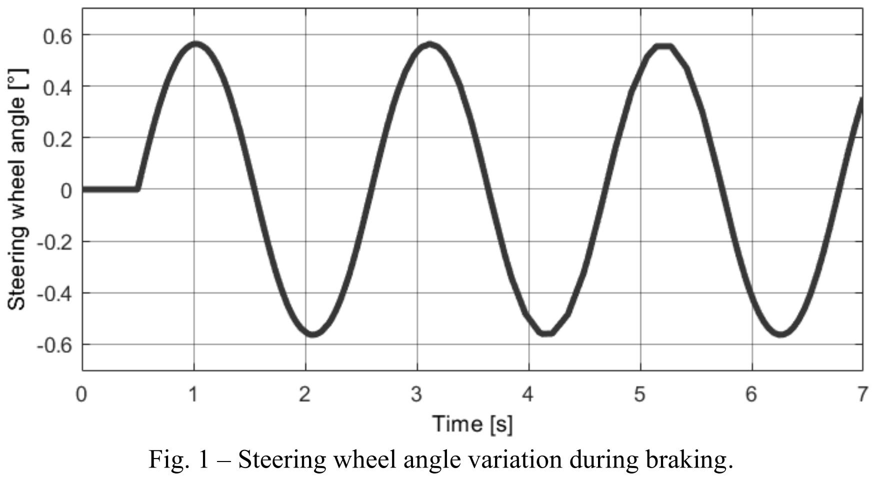

Fig. 1 – Steering wheel angle variation during braking.

The variation of the vehicle speed is the main parameter that must be study in order to estimate the stopping distance, the deceleration, the yaw rate and the slip ratio. As it can be noticed in Fig. 2, this parameter will be investigated by simulation procedures and the variation of it is expected to have significant impact on other parameters (Nguyen and Luong, 2023).

Fig. 3 presents the subsystems used from MATLAB Simulink-Antilock brake system (https://www.mathworks.com/help/hydro/ug/antilock-braking-system-il.html), extracting only the corresponding subsystem that is used by the secondary braking. The variation of wheel speed is calculated in accordance with the vehicle speed, while the dynamic behaviour of the vehicle and of the wheels are established by simulation using blocks parameters.

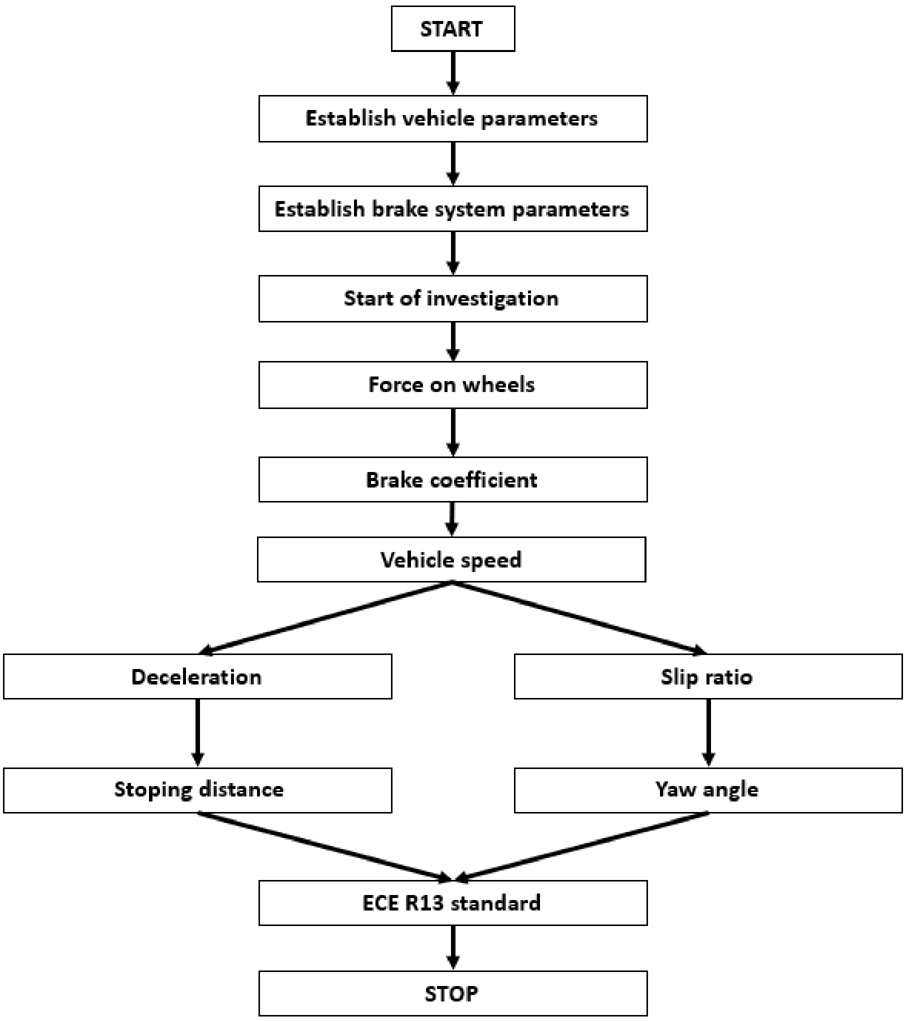

Fig. 2 – Methodology to evaluate the performance of a brake.

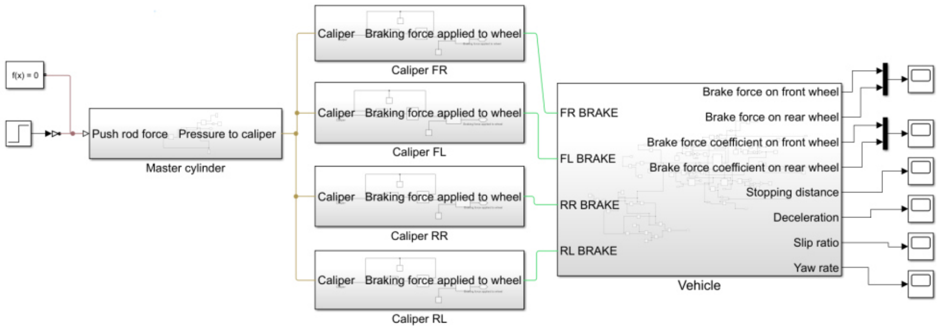

Fig. 3 – Blocks parameters used for investigation (https://www.mathworks.com/help/hydro/ug/antilock-braking-system-il.html).

According to ISO 611-2003 (ISO, 2003) and ISO/TR 13487-1997 (ISO, 1997), the Eq. (3) is used to calculate the variation of deceleration and the Eq. (4) is used to calculate the variation of the stopping distance,

$$a = dv/dt , \tag{3}$$

$$d = \int vdt , \tag{4}$$

where t is the time in seconds.

The braking force coefficient of a wheel, presented in the Eq. (5), is the braking force divided by the dynamic wheel load. For the case study, the braking force on the wheel and the dynamic wheel load are extracted from the vehicle body, routing NF (normal force for front) and NR (normal force for rear) from the Simulink model (Andrew, 2014),

$$BFC = F/N , \tag{5}$$

where F is braking force on the wheel [N] and N is dynamic wheel load [N].

The braking force on the wheel can be extracted from the Matlab&Simulink, by routing a force sensor block to each wheel. This parameter is the brake force applied by the brake system on each wheel, under normal braking conditions on the road (Andrew, 2014).

3. Case study and results

The case study refers to a Volkswagen Passat and the simulation starts within the fully load vehicle hypothesis.

The first step is to establish vehicle parameters and then the brake system parameters. Table 1 presents the parameters to be used for the vehicle and Table 2 presents the parameters that describe the wheel. The initial speed is chosen from ECE R13, type 0 test, with the value of 60 km/h. The vehicle decelerates on a straight road having a high friction coefficient, of 0.9, according to ISO 8608-1995 (ISO, 1995). This value represents the situation that occurs during the friction of a rubber wheel, on dry asphalt.

Table 1

Parameters of the vehicle

Parameters	Value	Unit
Mass	2300	kg
Distance from CG to front	1300	mm
Distance from CG to rear	1400	mm
CG from the ground	0.3	m
Frontal area	2.3	m^2
Drag coefficient	0.3	-
Moment of inertia	5300	kg/m^2
Front suspension stiffness	35000	N/m
Front suspension damping	4200	N/(m/s)
Rear suspension stiffness	32000	N/m
Rear suspension damping	3800	N/(m/s)
Initial speed	60	km/h

Table 2

Parameters of the wheel

Parameters	Value	Unit
Rolling radius	0.32	m
Friction coefficient	0.9	-
Inertia of wheel	2	kg/m^2
Rolling resistance coefficient	0.015	-

Next, the parameters of the brake system are presented. The data regarding the calipers are extracted from Volkswagen Passat datasheet and the data regarding the master cylinder and the pedal ratio are chosen from the book, Braking of Road Vehicles (Andrew, 2014), by taking into consideration average values.

Table 3

Parameters of the brake system

Parameters	Value	Unit
Master cylinder diameter	22	mm
Pedal ratio	4.3	-
Force applied to pedal	500	N
Front calipers diameter	60	mm
Rear calipers diameter	41	mm
Diameter of disk front	312	mm
Diameter of disk rear	282	mm
Coefficient of friction pad-disk	0.45	-

Simulation is performed by having in view several hypotheses: the vehicle will start braking at the time t = 0.5 s, after the initial time, by hard braking and maximum braking torque. As it can be noticed in Fig. 4, the braking force for front wheel is Ff = 2445 N and for the rear wheel is Fr = 1052 N.

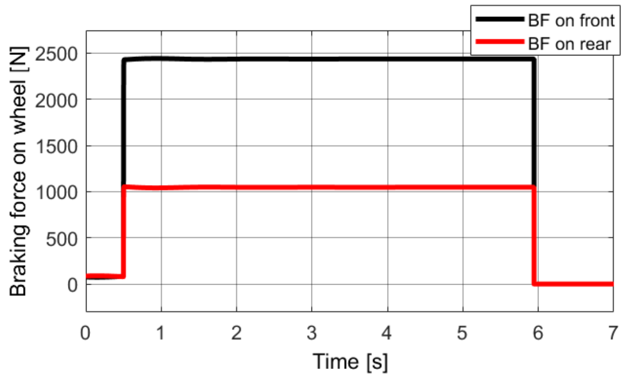

Fig. 4 − Braking force on each wheel.

The braking force on the front wheel represents 69.9% of the total braking force, while the braking force of the rear represents 30.1%. These results show that the distribution of braking forces does not exceed the optimal distribution for this type of vehicle, which should be 75% for the front wheel and minimum 25% for the rear wheel (Trzesniowski, 2023).

The next parameter to be studied is the brake force coefficient (BFC), which offers information on the dynamic friction coefficient at the highest point of the graph from Fig. 5. This parameter is simulated for both wheels (rear and front). The results are according to ECE R13 regulations, which stipulates that the BFC for the rear wheel must not exceed the BFC for the front wheel, during braking.

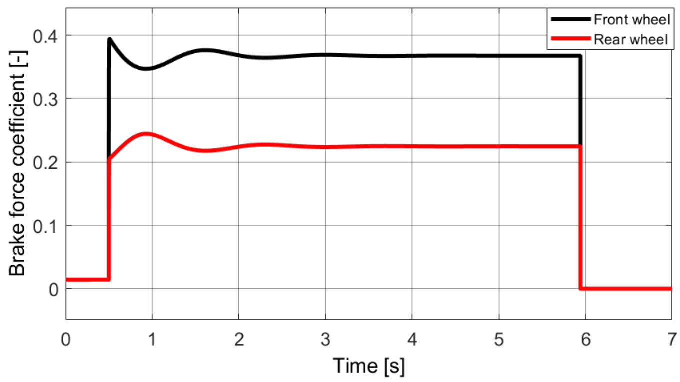

Fig. 5 – Brake force coefficient (BFC) for the front wheel and for the and rear wheel.

The representation of graphs does not start from the 0 value, because within 0 to 0.5 seconds time interval, the car is in deceleration without actuation of the braking system. For the front wheel, the maximum value of BFC is 0.388 and for the rear wheel, the maximum value is 0.23. These values represent the dynamic friction coefficient between the wheel and the road surface. It can be observed from Fig. 5, that during the braking period, the front wheel graph is above the rear wheel graph.

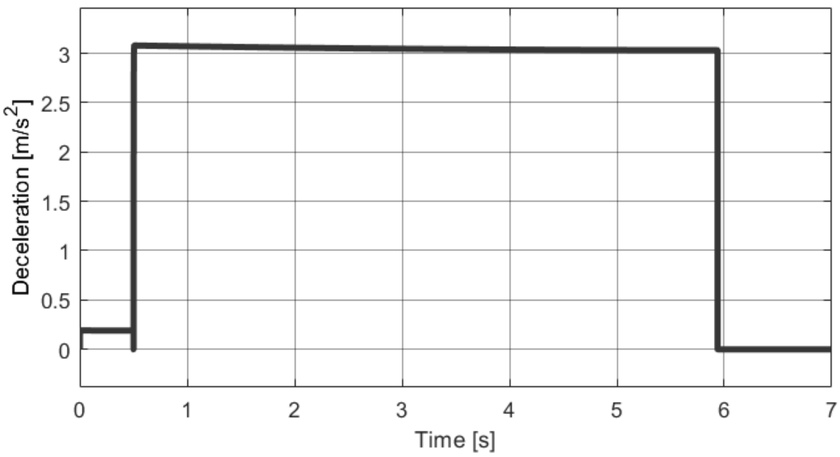

Fig. 6 – Variation of the deceleration of the vehicle.

The next studied parameter is the vehicle deceleration, during braking. According to ECE R13 regulations, the maximum deceleration value must be at least 2.5 m/s^2. In Fig. 6, the graph of deceleration of the vehicle indicates a maximum value of 3.08 m/s^2. The graph starts from 0 value because the deceleration has two different types of dynamic braking, one from time 0 to 0.5 seconds, namely the impact of deceleration of the free rolling car and the other from time 0.5 seconds to the moment when the car is stopped, namely the impact of the actuation of the braking system.

According to ECE R13, the stopping distance for the chosen case-study must not exceed 65 m, when the speed of vehicle is 60 km/h. The graph from the Fig. 7 indicates 53.25 m, by taking in consideration that the braking period starts at 0.5 seconds.

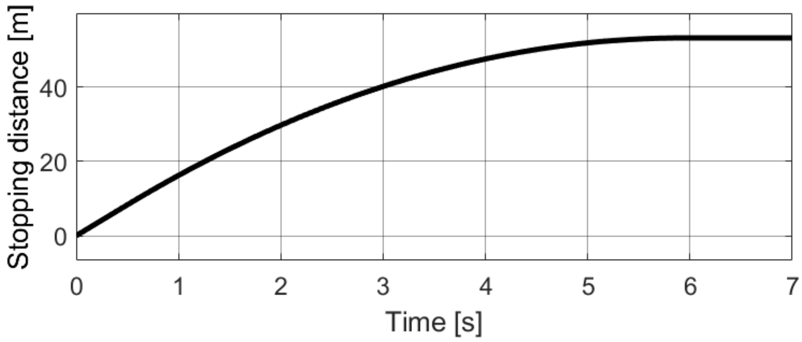

Fig. 7 – Variation of the distance travelled during braking.

Next parameter to be studied is the longitudinal slip ratio of the vehicle, presented in Fig. 8. This parameter should not reach the 100% value, otherwise the control of the vehicle is difficult, because the wheel is blocked. In the present case study, the slip ratio is 8.909%, which means that there is no problem regarding the control of the vehicle.

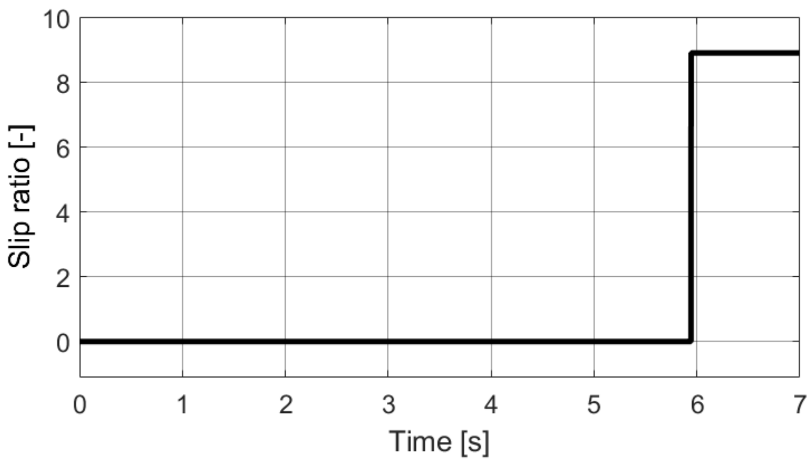

Fig. 8 – Variation of the slip ratio.

The yaw rate is another important parameter, necessary to evaluate the stability of the vehicle. For an adequate maneuverability of the car, this parameter should be 0, therefore a value close to 0 represents good maneuverability. For the case study, Fig. 9 presents the variation of the yaw rate during braking and the maximum value is 14.56 rad/s. After the braking period is over, the yaw rate becomes 0, which indicates good maneuverability (Raveendran *et al*., 2020).

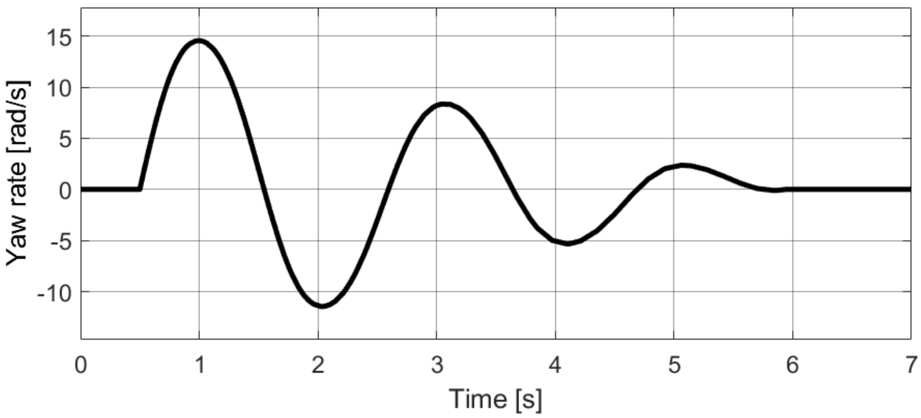

Fig. 9 – Variation of the yaw rate.

4. Conclusions

The present paper develops a methodology to investigate the performance of the secondary brake, by studying the dynamics behaviour, the brake bias, the stability and the control during braking. The results of the proposed methodology are discussed by verifying, if the requirements of ECE R13 regulations are achieved.

By using Matlab&Simulink, for the case study, a Volkswagen Passat vehicle moving by 60 km/h before braking, important parameters were evaluated: the stopping distance, the maximum deceleration, the maximum braking forces. Results indicate that the brake force coefficient, the slip ratio and the yaw rate reach values required by ECE R13. Moreover, the variation of yaw rate and slip ratio have a minor influence on the stability and the control of the vehicle.

Further investigation has in view, the increase of the vehicle speed, modification of the road profile, to compare results with the present study.

REFERENCES

Andrew D., *Braking of Road Vehicles*, Elsevier, 2014.

Aripin M.K., Yahaya M.S., Kumeresan A.D., Kumeresan A.D., Peng K. Hamzah N., Ismail M. F., *A review of active yaw control system for vehicle handling and stability enhancement*, International Journal of Vehicular Technology, 2014, 437515 (2014).

Dieter S., Manfred H., Roberto B., *Vehicle Dynamics Modeling and Simulation*, University of Duisburg-Essen, Duisburg, 2014.

Hans B.P., *Tyres and Vehicle Dynamics second edition*, Delt University of Technology, 2006.

Lei Z., Qi W., Jun C., Zhen-Po W., Shao-Hua L., *Brake-by-wire system for passenger cars: A review of structure, control, key technologies, and application in X-by-wire chassis*, eTransportation, 18, 100292 (2023).

Mathworks, *Matlab&Simulink version R2023b*, (2023). (https://www.mathworks.com/help/hydro/ug/antilock-braking-system-il.html)

Nguyen T.T., Luong V., *Modeling to study the braking efficiency of the electric vehicle,* Materials Today: Proceedings (2023).

Rajesh R., *Vehicle Dynamics and Control second edition*, University of Minnesota, Minneapolis, 2012.

Raveendran R., Devika K. B., Patil H., Subramanian S.C., *Yaw Angle Control of Heavy Commercial Road Vehicle with Faulty Brake*, IFAC-PapersOnLine, 53(1), 410-415 (2020).

Reza N.J., *Vehicle Dynamics Theory and Application*, Manhattan College, 2008.

Trzesniowski M., *Suspension system*, Springer, 2023.

United Nations Economic Commission for Europe, *European Commission, Regulation No 13 of the Economic Commission for Europe of the United Nations (UN/ECE) – Uniform provisions concerning the approval of vehicles of categories M, N and O with regard to braking [2016/194],* OJ L 42 2016, p.1.

ISO, *ISO 8608- Mechanical vibration- Road surface profiles- Reporting of measured data*, *1995.*

ISO, *ISO/TR 13487-Braking of road vehicles-Considerations on the definition of mean fully developed deceleration*, *1997.*

ISO, *ISO 611- Road vehicles- Braking of automotive vehicles and their trailers-Vocabulary*, *2003.*

INVESTIGAREA PERFORMANȚELOR UNEI FRÂNE SECUNDARE

(Rezumat)

În vederea creşterei siguranței în funcționare a autovehiculelor, sistemele de frânare actuale au frână secundară, capabilă să dezvolte forță de frânare pe roată, atunci când sistemul electric sau componentele sistemului ABS prezintă defecțiuni. Frâna secundară se bazează pe un cilindru principal (pistonul acționat de conducător și carcasa pistonului) și are o anumită configurație specifică fiecărui tip de sistem. De exemplu, pentru sistemul “one box”, cilindrul principal și ABS sunt încorporate într-o singură unitate, spre deosebire de soluțiile clasice la care cele două elemente sunt separate.

Lucrarea prezintă o metodologie de testare a eficienței frânei secundare a sistemului de frânare, utilizând facilitățile programului Matlab-Simulink. Scopul testării este de a verifica dacă soluția tehnică investigată este în conformitate cu standardul ECE-R13 și de a verifica dacă gradul de manevrabilite și controlul sunt adecvate. Este prezentat un studiu de caz. Studiul se referă un vehicul, Volkswagen Passat, complet încărcat având greutatea de 2300 kg și viteza inițială de 60 km/h. Rezultatele indică faptul că vehicolul

se oprește după ce străbate distanța de 53,25 m, având o decelerație de 3,08 m/s^2, forța de frânare pe față este de 2445 N, forța de frânare spate este 1052 N. Graficele indică că valoarea coeficientul forței de frânare are o variație adecvată pe durata frânării, iar raportul de alunecare al roților și unghiul de rotire sunt parametri ce nu au o influență majoră asupra gradului de control și manevrabilitate, în timpul frânării.